Enhanced Condensation for Organic Rankine Cycle

1st Quarterly Progress Report

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1. Back Ground

Generating electricity from low grade heat sources has attracted attention due to rising fuel price and increasing energy demand. The organic Rankine cycle (ORC) system is the most practical solution among technologies developed for low grade heat recovery. However, the efficiency of a typical small scale ORC is 10% or less. Most energy loss in the ORC is attributed to thermodynamically irreversible heat transfer processes occurring in its heat exchangers: the evaporator and condenser. In particular for waste heat recovery ORCs, economical success is mainly determined by effectiveness of the condenser because, while their heat source is provided at no cost, heat rejection accounts for most of operation cost. Almost half of total cost for operation and maintenance of an ORC system can stem from its condenser. We investigate and demonstrate heterogeneous condensing surfaces that potentially reduce the irreversibility during the condensation of organic fluids.

2. Progress Report

We have made progress during the reporting period (January 1 - April 30) and progress activities are described below.

Task 1: Model Development

The key indicator of condenser performance is the rate of condensation heat transfer. It is mathematically expressed by Newton's Law of Cooling; the rate of heat transfer Q is proportional to the condensing surface area A and the temperature difference between the vapor of organic fluid entering to the condenser and the coolant, $T_{\text{vapor@cond}}$ - T_{coolant}:

$$Q \sim A(T_{\text{vapor@cond}} - T_{\text{coolant}})$$

A factor called the heat transfer coefficient h turns the proportional relation above to:

$$Q = hA(T_{\text{vapor@cond}} - T_{\text{coolant}})$$

It represents how effectively the condenser transfers heat at a given surface area and temperature difference. The heat transfer coefficient h can be affected by a number of variables including the thermal properties of the working fluid, the geometry of the condensing surface, and the interactions between vapor and liquid and between liquid and solid, where the solid and the liquid are the condensing surface and the condensate, respectively. In this task, the solid-to-liquid interfacial phenomena and its impact on the condensation performance will be analytically studied.

During the reporting period, literal review was conducted for analytical studies on organic fluids. The most commercially available and applicable organic fluid for industrial condensers used to be chloro-fluorocarbons (CFC) refrigerants. Over the past few decades its environmental concern (depletion of the ozone layer) led to switch to hydro-fluorocarbons (HFC). These refrigerants are commonly used in air conditioners and refrigerators. Out of all the refrigerants R- 134a is becoming very popular in many applications. A research group has conducted a wide range of tests on R- 134a refrigerant to measure the condensation heat transfer coefficients and pressure drop when flowing in a small circular tube [1]. Unlike water, refrigerants or organic fluids do not readily condense dropwise because they have high wettability when compared to water. Discussed below are innovative techniques to overcome this challenge and to enhance the condensation heat transfer coefficient.

One of the techniques to enhance the condensation heat transfer coefficient is to increase the heat transfer surface area by decreasing the amount of film that forms on the condensing surface. This can be done by having the extended surface on the horizontal tubes within the shell and tube heat exchanger [2]. In this particular case the authors discuss shell-side condensation of pure vapor on the outside of horizontal tubes. The physical phenomenon which takes place is called Gregorig effect. The effect explains that the liquid meniscus at the top of the fin is convex whereas the base is concave, which results in the pressure difference. Pressure in liquid is greater at the fin top and lower at the fin base in comparison with the pressure in the vapor. The pressure gradient is the main reason for reducing the film thickness. Fluids with high wettability might be retained between the fins at the bottom of the tube. For this reason, drainage strip technique is used for fluids with high wettability. Porous strip that is attached to the base pulls the condensate into its pores creating low pressure region within the liquid at the base of the tube. Pores usually vary from 0.02 to 0.05 mm. Experimental investigations show noticeable improvement of heat transfer enhancement over smooth tube for R-22 refrigerant condensation [3].

Electrohydrodynamic (EHD) technique is based on electrically enhanced condensation by the force generated by an electric field. This effect can be achieved either by DC or AC low current with high voltage. The fluid flow is in a dielectric fluid medium between a charged and a receiving electrode [4]. The average heat transfer coefficient of refrigerant R-134a obtained with EHD is higher than without EHD. The EHD force pushes the refrigerant vapor having low dielectric permittivity to the inside tube surface. The liquid phase of refrigerant having high dielectric permittivity is pulled towards the electrode surface. This phenomenon generates the secondary fluid motion inside the liquid film, which helps in reducing the condensate thickness at the inside tube wall. The maximum enhancement in condensation heat transfer is approximately 30%.

Enhancement of condensation heat transfer is achieved in a plate heat exchanger with herringbone-type enhanced surface [5]. The inner surface of a plate heat exchanger is enhanced with cross grooved tubes, which are similar to micro fins. They are either rectangular or trapezoidal shaped with a depth ranging from 0.1 to 0.25 mm, a pitch from 0.4 to 0.5 mm and a spiral angle from 100 to 300 degrees. Their special shape enables to achieve more heat transfer coefficients and less pressure loss compared to conventional surface and micro finned surfaces. With the increase in surface roughness and in the density of nucleation sites for vaporization, the cross grooved tubes act as drainage for the condensate, which results in the enhancement of the condensation heat transfer.

Task 2: Design and Construction of Testing Apparatus

Design for a condensation experiment apparatus has been completed in a SolidWorks environment as shown in Figure 1. The testing setup consists of the condensing chamber, evaporator, condensing surface and heat conduction rod, and cooling chamber.

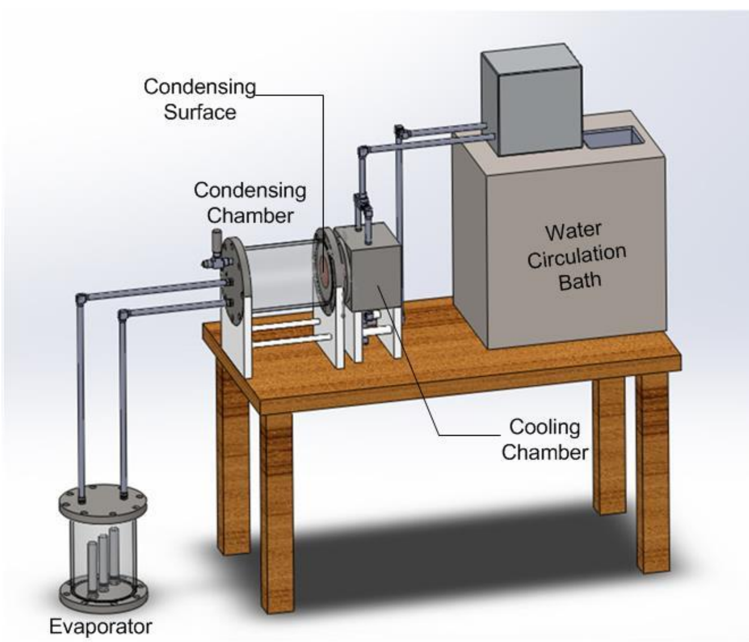
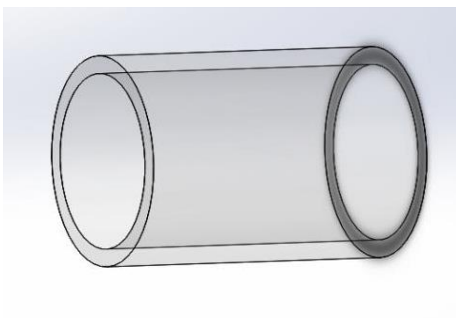


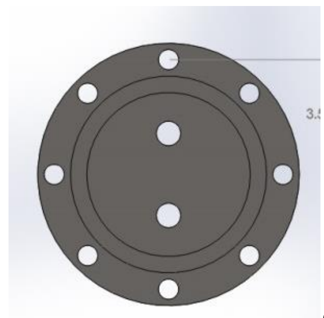
Figure 1: Design of condensation experiment apparatus

Evaporator

A one third of an inch thick, 6 inches outer diameter and 8 inches long borosilicate glass cylinder is chosen as the wall of the boiling chamber (Figure 2a). Borosilicate is well known for its low coefficient of thermal expansion (3×10^{-6} 1/K). Owing to the low coefficient of thermal expansion it is highly resistant to thermal shock. Two stainless steel (type 303) disks of thickness ¾th of an inch and the diameter of 8 inches are used to close the two open ends of the cylinder (Figure 2b). The two stainless steel disks have 1/8th inch deep and ½ inch wide groove. A neoprene gasket of thickness 1/8th of an inch, and ½ inch wide, and 5 inches inner diameter sits in the groove. The stainless steel disks will be bolted together using commercial nuts and bolts. The stainless steel disk, from which the hoses are connected to the condensing chamber, has two ¾th of an inch NPT male hose fitting welded. The organic vapor flows into the condensing chamber through a hose, and the condensate flows back to the evaporator through the second hose. The stainless steel disk on the other end of the borosilicate glass has three 1/4th of an inch FNPT fittings welded so that we can screw in cartridge heaters.



(a)



(b)

Figure 2: The borosilicate glass cylinder (a) and the stianless steel disk for the evaporator (b)

Condensing chamber

The condensing chamber wall is also made of borosilicate glass. The wall is a cylindrical borosilicate glass which is one third of inch thick and 8 inches long and outer diameter of the cylinder is six inches. Two stainless steel disks (type 303) are used to close the cylinder ends. The stainless steel disks have grooves in similar fashion as in the evaporator. The gasket fits in the groove and when the two stainless steel disks are bolted. The stainless steel disk has two 3/4th inch NPT male hose fitting, which are welded so that we can connect hoses to condensing chamber from the boiling chamber. The same disk has 3/8 inch FNPT fitting welded so that a pressure transducer can fit in. There is also ¼ inch FNPT fitting welded for a thermo couple that measures the organic vapor temperature. The other stainless steel disk that can be seen in Figure 3 has a 3 inch diameter hole from the center so as to incorporate a copper block, which would be on the same plane with the inner side of the disk. Also on the outer side of the stainless steel disk there is a groove which is made from the center of the disk, with an inner diameter 3 ½ inch and outer diameter of the groove is 3 3/8 inches. The depth of the groove is 1/8th of an inch. The groove is made to fit in an o-ring between the copper block and the stainless steel disk.

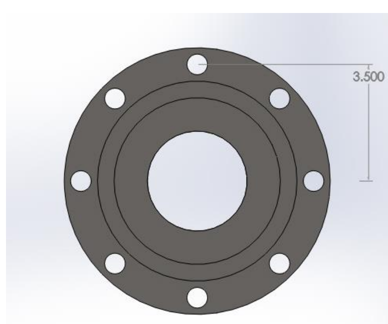


Figure 3: The stainless steel flange with groove and socket for copper block

<u>Cooling chamber</u>

The cooling chamber is made out of stainless steel type 304. The 6" x 6" x 4" box-shaped chamber will be filled with the coolant that is supplied by the chiller pump. The front face of the cooling chamber has a hole of 3-inch diameter for the copper rod, which acts as a thermal flow path connecting the heating chamber with the cooling chamber. The heat conduction copper rod extrudes in 2 inches deep into the cooling chamber, so as to reject the heat from the organic vapors in the condenser. The front face of the cooling chamber has four holes with 1/8 inches deep and with diameter of 0.4 inches, which are drilled in a circular pattern. Socket head bolts are chosen, and they are arranged in such a way that the bolt head goes into the holes. A 3/8 inches stainless steel disk which has identical holes to those on the front face of the cooling chamber is then screwed onto the front face of the cooling chamber as shown in Figure 4. The heating chamber stainless steel disk has threaded holes on its outer surface, which would incorporate these bolts. By turning the bolts the heating chamber and the cooling chamber would be tighten the copper rod, which provides leak proof configuration in both the heating chamber and the cooling chamber at the same time. The top and bottom faces of the cooling chamber have two holes each of ¼ inch, so that a hose fitting can be welded on. Hoses are connected to the chiller pump that supplies a constant temperature coolant.

<u>Heat conduction copper rod</u>

The copper rod with 3.5 inches diameter and length of 4.5 inches is machined in a lathe in such a way that one end of the copper rod has 3 inches diameter for 2 inches long and the other end has 3 inches diameter for 0.75 inches long. The copper rod is shown in Figure 5.

The assembly of the condenser and the cooling chamber will look like Figure 6 shown below. In the picture it is clearly seen that the condenser and the cooling chamber are attached and the copper rod is tightened with the bolts.

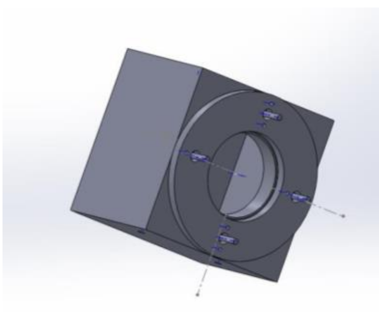


Figure 4: Cooling chamber welded with stainless steel disk and bolts

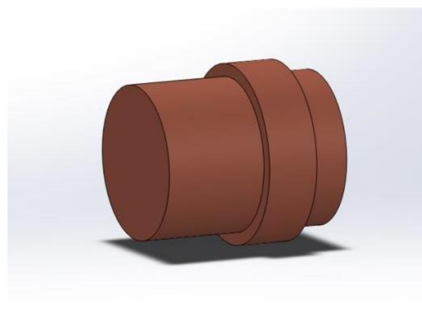


Figure 5: Copper rod acts as a thermal path connecting the condenser and the cooling chamber.

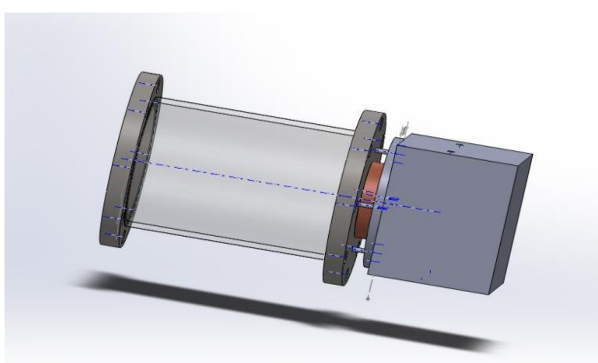


Figure 6: Condenser and cooling chamber assembly

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